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NASA TN D-1583

N63-13965
Code-1

NASA TN D-1583

TECHNICAL NOTE

D-1583

SUBCOOLED BOILING HEAT TRANSFER UNDER FORCED CONVECTION IN A HEATED TUBE

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

WASHINGTON March 1963

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SUMMARY

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Single- and two-phase heat-transfer data were obtained by using distilled water flowing through an Inconel X resistance-heated tube. The nonboiling data were correlated with a Colburn-type equation that was modified to include the boiling phenomena by means of three significant parameters obtained by dimensional analysis from basic considerations. Comparable heat-transfer data from reference sources, covering a broad range of conditions, were used in the development of the correlation. The reference data and the limited data obtained from the experimental program extended the generality of the correlation to cover a range of pressure from 16 to 2000 pounds per square inch, heat flux from 0.026 to 56.0 Btu per square inch per second, fluid velocity from 1.33 to 204 feet per second, and subcooling from 6° to 336° F. Liquid-ammonia data were included to demonstrate the applicability of the correlation to fluids other than water.

Comparisons were made between wall temperatures at the incipience of boiling as predicted by an analytically derived equation and the experimental data.

INTRODUCTION

Emphasis has previously been placed on obtaining engineering correlations of experimental data from both pool and forced-flow boiling systems. Unfortunately, a basic approach at obtaining an understanding of boiling mechanisms is hampered by the complex interaction of the many parameters involved, and analysis becomes difficult. Many unknowns such as the dependence of heat flux or surface conditioning and the statistical nature of bubble growth are difficult to evaluate. Equations presented in the literature are usually derived from limited data by using correlating techniques that do not include all the significant parameters.

The confusion that exists in connection with pool boiling is shown in reference 1 by a partial list of correlations obtained since 1952. These equations are reliable for individual sets of data but not for general use. Calculations have shown that deviations of heat flux can vary by a factor of 2 or more.

The present investigation is concerned primarily with the forced-flow system. The added complexity of the fluid velocity on the ebullition process makes analysis more difficult than would be expected for the pool-boiling system. Correlations presented in the literature are, therefore, either strictly empirical or based on some semiempirical method. References 2 to 12 contain a partial listing of such correlations, which are subject to the limitations existing in pool boiling. Correlations that work well for one set of data do not necessarily fit data from other facilities even for similar test conditions.

Although it is not expected that a unique set of data could ease the confusion that exists, it is felt that reliable heat-transfer data are required to provide the tools for obtaining a more general understanding of the boiling phenomena. An experimental investigation was initiated with the expectation of using the data obtained to develop a more general type of engineering correlation.

Single- and two-phase heat-transfer data were obtained by using distilled water flowing through an Inconel X resistance-heated tube. The test section was 6.5 inches long and had an inside diameter of 0.311 inch. The system variables included a range of pressure from 37 to 179 pounds per square inch absolute, velocity from 3.8 to 12.5 feet per second, heat flux from 0.37 to 1.60 Btu per square inch per second, and subcooling from 180° to 263° F.

The nonboiling heat-transfer data obtained were first correlated using a Colburn-type equation made up of a group of dimensionless parameters. The boiling phenomena were then included by modification of this convection equation with three significant parameters obtained by dimensional analysis from basic considerations. The initial parameter used relates the volumetric rate of vaporization of the liquid to the fluid velocity along the heated surface. This relation was presented in reference 8 as a correlation of boiling data limited to a fixed pressure level and, when substantiated, was used as the basis for the present correlation. Two additional parameters were used to include the effect of pressure and degree of subcooling in the final correlation. The development of the correlation included forced-convection boiling heat-transfer data obtained from the literature, which covered a wide range of fluid properties and flow conditions. Liquid-ammonia data were shown to fit the correlation.

Experimentally determined wall temperatures were compared with calculated wall temperatures by employing an analytically derived equation (ref. 13) that predicts the incipience of boiling.

EXPERIMENTAL EQUIPMENT

Flow System

The flow system, which employed an explusion bag for obtaining steady fluid flow through the test section, is shown schematically in figure 1(a). Distilled water was contained in a neoprene-type bladder installed in the tank upstream of a flow-measuring orifice. Nitrogen gas, at controlled pressures, was introduced between the inner wall of the tank and the outer surface of the bladder to force the fluid through the flow system.

Mixing chambers, each consisting of a system of baffles, were installed before and after the test section to eliminate temperature stratification in the fluid bulk. A system of valves and regulators controlled the flow rate and pressure level throughout the apparatus. The fluid discharged into the collector tank, which also contained an expulsion bladder for recycling. The piping apparatus was constructed entirely of stainless steel to minimize contamination problems.

Test Section

A schematic drawing of the instrumented test section (fig. 1(b)) shows measuring stations for wall temperature, pressure, and voltage drop. Inconel X tubing with an inside diameter of 0.311 inch and a 0.012-inch wall thickness was employed. The resistance-heated portion of the test section was 6.5 inches long. Pressure tubes, voltage taps, and iron-constantan thermocouples were silver soldered to the tube.

A 9000-watt alternating-current generator supplied power to heat the test section through two electrodes brazed to the outer wall of the tube. The power input was controlled by a variable transformer. The test section was electrically insulated from the rest of the flow system and was wrapped in Fiberglas to minimize ambient heat loss.

Instrumentation

Bulk temperature and pressure were measured in the mixing chambers located at the inlet and the exit of the test section. The test section was instrumented with 12 thermocouples made of 28-gage iron-constantan wires installed in two rows along the length of the tube located 180° apart. The five pressure taps were made of 0.035-inch-outside-diameter stainless-steel thin-wall tubing. The five voltage taps were made of 28-gage copper wire .

All the basic data, including temperatures, pressure, flow rate, and alternating-current tube voltages, were converted to low direct-current voltage so that they could be recorded on a multichannel oscillograph.

EXPERIMENTAL PROCEDURE

The controlled variables for operating the test apparatus included system pressure, flow rate, and power to heat the test section. For fixed values of pressure and flow rate, data were obtained at discrete intervals of power input to the limitations of the power source. At each power setting sufficient time was allowed for the system to reach steady-state conditions before the data were recorded. The same procedure was repeated for a range of flow rate and system pressure. At low flow rates, the wall temperature limited the amount of electric power that could be dissipated in the tube.

The heat-transfer data covered a range of system pressure from 37 to 179 pounds per square inch absolute, fluid velocity from 3.8 to 12.5 feet per second, heat flux from 0.37 to 1.60 Btu per square inch per second, and subcooling from 180° to 263° F.

COMPUTATION PROCEDURE AND DATA PRESENTATION

Determination of experimental heat-transfer coefficients required local values of heat flux, inside surface wall temperature, and bulk temperature. The heat flux was obtained directly from measured values of current and voltage drop by the following equation:

$$q = 0.984\times10^{-3}\,\frac{EI}{A_i} \tag{1}$$

(All symbols are defined in appendix A.) Equation (1) denotes a uniform heat-flux distribution because of the insignificant change in electrical resistivity throughout the range of wall temperatures obtained. Verification of a linear voltage drop was made by five voltage taps spaced over the length of the test section.

Since the heat-transfer coefficient is based on heat flux from the inner surface of the tube, the measured outside wall temperatures were corrected for temperature drop through the wall. The following theoretical equation that assumes uniform internal power generation was obtained from reference 2:

$$T_i = T_o - K\,\frac{Q}{k} \tag{2}$$

where

$$K = \frac{r_o^2 \ln\dfrac{r_o}{r_i} - \left(\dfrac{r_o^2 - r_i^2}{2}\right)}{2\pi L\left(r_o^2 - r_i^2\right)}$$

The local bulk temperatures were obtained by assuming sensible heating of the fluid as indicated by the following equation:

$$T_{b,x} = T_{b,in} + \frac{Qx}{L\dot{m}c_p} \tag{3}$$

The second term on the right side of the equation is a measure of the temperature rise of the fluid caused by the heat input. This term was evaluated at each measuring station and added to the measured inlet bulk temperature to obtain the local bulk temperatures. The sensible heating assumption is correct for the single-phase heat-transfer data and can be accepted as valid for the two-phase heat-transfer data because of the high subcooling involved.

In order to eliminate uncontrollable end effects, the data presented were taken from the midportion of the test section. Table I lists the data and completed computations for temperature measuring station number 11, which was chosen as representative of typical data in the midportion of the test section.

CORRELATION PROCEDURE AND DISCUSSION OF RESULTS

Nonboiling

The nonboiling heat-transfer data obtained in the present investigation were correlated by using a Colburn-type equation (ref. 2) with fluid properties evaluated at film temperature T_f. The logarithmic plot of the results presented in figure 2 shows a data scatter of approximately 20 percent. The equation of the dashed line representing the correlation is

$$Nu_{calc} = 0.021\left(\frac{\rho_f V_b d}{\mu_f}\right)^{0.8}\left(\frac{c_p \mu}{k}\right)_f^{0.4} \tag{4}$$

References to calculated Nusselt number will imply use of equation (4) only when dealing with the experimental data presented herein. Nonboiling correlations presented in reference sources will be associated with their respective data.

Subcooled Boiling

Forced-convection boiling data from references 3, 4, 5, 10, and 11 along with the data obtained from the experimental program described herein were used to develop an effective correlation. The wide range of variables in the reference data (table II) increased its generality. The data were correlated by means of three significant parameters obtained by dimensional analysis from basic considerations. A nondimensional parameter presented in reference 8 as a limited relation of boiling data was used as the starting point of the present correlation. Two additional parameters were then determined to compensate for a subcooling effect (independent of pressure) and a pressure effect that were revealed by an evaluation of all the data.

The initial correlation of the experimental data is presented in figure 3, which shows a ratio of Nusselt numbers plotted against a dimensionless parameter. The ratio of Nusselt numbers is used consistently in all succeeding plots, and the development of the correlation is indicated by the changes in the parameters on the abscissa. The numerator of the ratio of Nusselt numbers is an experimentally determined Nusselt number based on local values of heat-transfer coefficients, and the denominator is a calculated Nusselt number based on equation (4) or any nonboiling forced-convection correlating technique specified in the references. The Nusselt number ratio remains at a value of unity for all nonboiling data, and it is greater than unity when boiling persists because of the increased heat-transfer coefficient in the experimental Nusselt number.

The parameter $q/\lambda\rho_v V_b$ depicts a correlation of boiling heat-transfer data obtained at a unique pressure level (ref. 8), and it was developed from a dimensional analysis of the basic heat-transfer mechanism. The existence of two distinct modes of heat transfer was assumed from the laminar transition layer along the wall to the bulk of the boiling fluid. The first mode was responsible for the amount of heat transfer by turbulent mixing as a result of the velocity gradient. The second mode was a measure of the heat transfer due to molecular mass transfer caused by bubbles departing from the heated surface. A detailed description of the analysis may be found in reference 8.

The parameter $q/\lambda\rho_v V_b$ was calculated by using the experimental data obtained in this investigation, and figure 3 depicts its limitations in affecting a correlation. Boiling data at a unique pressure level are on a line having a slope of 0.7 and increased pressures shift this line to the left. Both of these observations are resported in reference 9 along with a complete correlation based on two additional parameters to correct for changes in pressure level. The equation presented (ref. 9) does not correlate the present experimental data or the reference data used in this investigation. The unavailability of the data used to obtain this correlation makes explanations for this discrepancy difficult.

An examination of the available data revealed the existence of a subcooling effect, independent of pressure level, that must be compensated in any effective correlating procedure. Figure 4 is a plot of the correlating parameter applied in figure 3 with data from references 3 to 5 obtained at a pressure level of 2000 pounds per square inch absolute. The amount of subcooling is marked at each datum point, and dashed lines with a slope of 0.7 are drawn through nearly constant values of subcooling. The spread of these lines clearly indicates an effect of as much as two orders of magnitude for these data.

In order to compensate for the subcooling effect, a parameter obtained by a strictly empirical approach was determined from the experimental data. The reciprocal of the amount of subcooling raised to the 1.20 power $[1/(T_s - T_b)]^{1.20}$ proved to be an effective correlation. In order to maintain nondimensionality, the parameter was modified to include the heat of vaporization and the specific heat of the fluid $[\lambda/c_p(T_s - T_b)]^{1.20}$. This particular grouping had been obtained from a parametric evaluation of the heat balance in and out of a control volume containing a boiling fluid (ref. 7). The spread in the data of figure 4 was effectively eliminated when the data were replotted in figure 5 by including the parameter $[\lambda/c_p(T_s - T_b)]^{1.20}$ as an integral part of the correlating equation (fig. 5). The heat of vaporization was evaluated at saturation conditions and the specific heat at the mean temperature between saturation and local bulk.

The spread of data due to pressure (fig. 3) could effectively be eliminated by the inclusion of a parameter consisting of the ratio of vapor density to liquid density previously used in reference 9. An exponent equal to 1.08 was empericaily derived when the density ratio was evaluated at saturation conditions $(\rho_v/\rho_l)^{1.08}$.

The completed correlation presented in figure 6 includes the data obtained in the present investigation and the boiling heat-transfer data from five reference sources covering a broad range of conditions. Table II lists the range of pertinent variables. The data included a range of pressure from 16 to 2000 pounds per square inch, heat flux from 0.026 to 56.0 Btu per square inch per second, fluid velocity from 1.33 to 204 feet per second, and subcooling from 6° to 336° F. The spread of the data about the solid line in figure 6 shows the effectiveness of the correlation. Approximately 92 percent of 260 data points are within ±12 percent, as indicated by the dashed lines. Fifteen points from three runs in reference 4 are consistently plotted below the line in the lower portion of the curve and are not included in the evaluation of the data scatter. These points are a small fraction of the data used from that particular reference and appear to be inconsistent. The scatter in the upper portion of the curve is a result of the low subcooling involved since small errors in bulk temperature measurements can result in large deviations. The nonboiling region of the correlation exists for values of $(q/\lambda\rho_v V_b)[\lambda/c_p(T_s - T_b)]^{1.2}(\rho_v/\rho_l)^{1.08}$ less than 0.00162, which is the incipient boiling point. When boiling persists, the data are correlated by the following equation:

$$\frac{Nu_{exp}}{Nu_{calc}} = 90.0\left\{\left(\frac{q}{\lambda\rho_v V_b}\right)\left[\frac{\lambda}{c_p(T_s - T_b)}\right]^{1.20}\left(\frac{\rho_v}{\rho_l}\right)^{1.08}\right\}^{0.7} \tag{5}$$

In order to demonstrate the applicability of the correlation for fluids other than water, liquid-ammonia boiling heat-transfer data from reference 9 were applied to equation (5). The data covered a range of variables that includes pressure from 170 to 1174 pounds per square inch absolute, heat flux from 0.38 to 9 Btu per square inch per second, velocity from 3 to 85 feet per second, and subcooling from 37° to 187° F (fig. 7). The percent deviation is within the range of the water data presented in figure 6 except for the four points obtained at a pressure level of 170 pounds per square inch absolute.

The correlating equation (5) can only be applied to subcooled boiling. The parameter containing the degree of subcooling of the fluid becomes infinite when the bulk temperature approaches saturation conditions. Further studies are required to determine parameters suitable for correlating saturated boiling data.

Incipience of Boiling

A great deal of interest has been expressed in a method for predicting the conditions required for the incipience of boiling in a subcooled fluid. An analytical treatment of this problem, presented in reference 13, results in an equation that involves the cavity site and the thermodynamic state of the thermal layer. This equation cannot be solved directly because of the difficulty of obtaining the values of two constants. One of the constants is a function of the dimensions of the bubble site cavity. The other constant is the laminar sublayer thickness, which varies with stream velocity. If incipient boiling data are available, it is possible to calculate the value of the ratio of the two constants by using the equation of reference 13. With this ratio evaluated at a unique velocity, it is possible to predict the variation of wall temperature with pressure for that specific velocity. A check on the validity of this equation was made by using the experimental data obtained in this investigation. The incipient boiling data were obtained from figure 6 at the point where the Nusselt number ratio equals unity and the correlating parameter equals 0.00162. The calculation procedure is presented in appendix B. The results show a small difference between the experimental and calculated wall temperatures at the inception of boiling.

SUMMARY OF RESULTS

Single- and two-phase heat-transfer data were obtained by using distilled water flowing through an Inconel X resistance-heated tube. The nonboiling data were correlated by a modified Colburn-type equation within a 20-percent scatter. The subcooled boiling data correlated within ±12 percent by an equation, which included a unique parameter to compensate for changes in subcooling independent of pressure.

The generality of the correlation was increased by using boiling heat-transfer data obtained from reference sources. The range of variables effectively correlated included pressure from 16 to 2000 pounds per square inch absolute, heat flux from 0.026 to 56.0 Btu per square inch per second, fluid velocity from 1.33 to 204 feet per second, and subcooling from 6° to 336° F. Liquid-ammonia data obtained from the literature correlated readily within the scatter of the water data. Further studies should be made before an attempt is made to employ the boiling correlation to fluids other than those investigated.

Comparisons were made between wall temperatures at the incipience of boiling as predicted by an analytically derived equation and by the experimental data. The results show a small difference between analytical and experimental wall temperatures. The equation used has limited applicability because experimental data must be available to calculate constants that cannot be directly measured. These constants can be evaluated at a specific velocity and then used to predict the variation of wall temperature with pressure for that velocity.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, December 4, 1962

APPENDIX A

SYMBOLS

A	area
a	$2\sigma T_s/\lambda\rho_v$ (eq. (B1)), units are (ft)($^{\circ}$R) when δ in eq. (B1) is in ft
C_3	$1 + \cos\varphi$
c_p	specific heat at constant pressure
d	inside diameter of tube
E	voltage
h	heat-transfer coefficient
I	current
K	constant in eq. (2)
k	thermal conductivity
L	total length of test section
$\dot{m}$	mass-flow rate
Nu_{calc}	Nusselt number computed from modified Colburn-type equation by using film temperature to evaluate fluid properties
Nu_{exp}	experimental Nusselt number obtained from measured heat-transfer coefficient, hd/k
Pr	Prandtl number, $(c_p\mu/k)_f$
p	pressure
Q	heat flow
q	heat flux
Re	Reynolds number, $\rho_f V_b d/\mu_f$
r	radius of tube
T	temperature
T_f	$(T_{w,i} + T_b)/2$

V	velocity
x	distance to temperature station measured from beginning of heated portion of test section
β	contact angle
γ	angle of tangent to cavity mouth with respect to horizontal
δ	laminar sublayer thickness
θ_{wo}	$T_w - T_b$
θ_s	$T_s - T_b$
λ	heat of vaporization
μ	viscosity
ρ	density
σ	surface tension
φ	angle of bubble wall with respect to horizontal, $\gamma + \beta$

Subscripts:

b	bulk fluid
f	film
i	inner surface of test section
in	inlet
l	liquid
o	outer surface of test section
s	saturation
v	vapor
w	wall at incipience of boiling

APPENDIX B

INCIPIENT BOILING POINT

Experimental data obtained in this investigation were used to check the validity of an analytically derived method of predicting the surface temperature at the inception of boiling (ref. 13). The equation is

$$\theta_{wo} = \theta_s + \frac{2aC_3}{\delta} + \sqrt{\left(2\theta_s + \frac{2aC_3}{\delta}\right)\left(\frac{2aC_3}{\delta}\right)} \tag{B1}$$

The incipience of boiling can be obtained from equation (B1) if C_3 and δ are known. The quantity C_3 is a function of the shape of the bubble-site cavity. The quantity δ, which is the thickness of the laminar sublayer, is a function of stream velocity. Unfortunately, these values are not readily available.

If the incipient boiling point is experimentally known, it is possible to calculate the ratio δ/C_3 by equation (B1). This unique boiling point is readily obtained from the correlation presented in figure 6 at the point where the Nusselt number ratio initially departs from a value of unity. At this point, the value of the correlating parameter on the abscissa is 0.00162.

Four incipient boiling points were chosen; data were obtained at the same velocity but different pressures. The ratio δ/C_3 was calculated from one of these points and should remain constant as long as the velocity is constant (ref. 13). This ratio was then used to calculate the wall temperatures for the other three chosen points. The calculated and experimental wall temperatures at the inception of boiling were then compared. The computations showing the agreement between computed and experimental wall temperatures are shown in table III.

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TABLE I. - SINGLE- AND TWO-PHASE HEAT-TRANSFER DATA AT STATION 11

Run	Temperature				Pressure, P, lb/sq in. abs	Heat flux, q, Btu/(sec)(sq in.)	Mass-flow rate, $\dot{m}$, lb/sec	Bulk velocity, V_b, ft/sec
	Outside, T_o, °F	Inside, T_i, °F	Bulk, T_b, °F	Saturation, T_s, °F				
1172	196	185	81	271	41.7	0.398	0.221	6.71
1174	255	238	88	272	42.9	.632	.219	6.62
1175	290	265	85	273	43.8	.914	.234	7.12
1176	312	280	94	273	43.4	1.21	.237	7.22
1178	369	328	108	264	37.0	1.60	.217	6.63
1179	208	196	85	274	44.7	.411	.186	5.66
1180	279	261	89	277	46.8	.622	.188	5.74
1181	286	262	95	278	47.6	.912	.184	5.61
1182	338	307	105	280	49.3	1.22	.180	5.49
1183	376	335	110	280	49.8	1.59	.189	5.75
1184	381	340	112	282	51.0	1.60	.182	5.56
1190	233	222	82	270	41.3	.409	.144	4.38
1191	291	275	83	273	43.6	.630	.164	4.99
1192	298	274	91	275	45.1	.913	.164	5.00
1193	356	324	101	276	46.4	1.24	.162	4.94
1195	380	340	105	278	48.1	1.56	.164	5.00
1196	246	235	80	283	51.5	.423	.130	3.97
1197	291	274	64	282	51.1	.641	.141	4.29
1198	322	298	92	265	53.2	.930	.139	4.23
1199	361	330	101	268	55.7	1.24	.141	4.30
1200	388	348	111	288	56.1	1.59	.139	4.25
1202	165	154	86	312	80.8	.400	.339	10.31
1203	211	194	91	313	81.2	.630	.354	10.78
1204	257	232	94	316	85.9	.899	.359	10.94
1205	311	279	94	319	88.4	1.23	.364	11.09
1206	361	321	102	310	77.8	1.57	.336	10.25
1228	187	175	84	294	60.6	.401	.273	8.30
1229	181	170	83	343	119.0	.399	.269	8.15
1230	238	220	88	344	120.3	.646	.268	8.15
1231	288	264	83	345	126.1	.907	.274	8.33
1232	366	335	89	342	117.2	1.230	.253	7.70
1233	386	346	94	327	99.0	1.570	.253	7.71
1235	210	198	81	330	102.5	.402	.189	5.75
1236	281	265	88	324	96.2	.619	.176	5.36
1258	363	339	96	330	104.8	.921	.190	5.79
1260	388	358	104	336	110.5	1.200	.192	5.86
1261	413	375	109	336	112.6	1.560	.182	5.56
1262	229	217	85	339	116.4	.419	.166	5.05
1263	231	219	87	341	119.0	.412	.162	4.93
1264	313	296	93	336	112.6	.635	.154	4.69
1266	363	340	96	327	98.2	.894	.162	4.94
1267	378	347	102	325	94.4	1.220	.164	5.00
1268	403	364	108	326	97.3	1.550	.167	5.10
1269	248	237	98	325	97.8	.415	.138	4.20
1270	329	312	103	327	98.6	.628	.138	4.21

Run	Temperature				Pressure, P, lb/sq in. abs	Heat flux, q, Btu/(sec)(sq in.)	Mass-flow rate, $\dot{m}$, lb/sec	Bulk velocity, V_b, ft/sec
	Outside, T_o, °F	Inside, T_i, °F	Bulk, T_b, °F	Saturation, T_s, °F				
1271	372	349	113	327	100.3	0.305	0.141	4.32
1273	357	357	117	328	99.5	1.230	.143	4.36
1274	420	392	126	329	101.2	1.350	.143	4.39
1275	167	155	89	316	95.0	.415	.353	11.57
1276	206	190	93	343	119.5	.633	.385	11.52
1277	255	231	98	336	123.7	.392	.393	11.95
1278	304	272	100	341	122.4	1.220	.399	12.17
1279	349	309	101	345	126.2	1.360	.401	12.23
1284	192	180	90	352	139.9	.416	.266	8.08
1286	236	218	93	353	141.0	.633	.266	8.10
1287	294	270	92	354	143.7	.353	.223	6.20
1288	360	329	100	344	148.0	1.210	.270	8.24
1289	412	373	106	356	148.0	1.370	.274	8.37
1290	222	210	93	365	162.0	.417	.178	5.45
1291	285	268	114	366	163.7	.626	.151	5.54
1292	349	322	97	366	165.0	.395	.144	4.67
1293	413	383	104	367	165.0	1.200	.150	5.50
1294	439	402	111	367	168.5	1.340	.150	5.51
1295	241	230	94	366	174.3	.405	.143	4.36
1296	318	302	100	367	176.9	.625	.146	4.47
1297	394	372	103	372	177.7	.900	.146	4.45
1298	424	394	110	373	178.8	1.210	.146	4.46
1307	442	404	122	359	145.8	1.54	.138	4.23
1309	157	147	84	344	124.3	.371	.365	11.33
1310	197	173	89	345	126.6	.590	.359	11.34
1311	239	216	[illegible]	348	124.8	.560	.384	11.89
1312	288	257	88	349	128.9	1.17	.399	12.14
1313	345	305	93	349	132.9	1.54	.409	12.41
1314	294	271	88	354	132.5	.806	.256	7.79
1315	347	317	93	353	130.4	1.17	.266	8.16
1316	410	372	100	354	135.8	1.50	.211	6.27
1322	348	326	103	360	172.4	.892	.190	5.60
1323	404	375	111	361	163.6	1.15	.158	4.65
1324	437	400	121	361	154.1	1.50	.155	4.76
1325	318	299	106	364	161.0	.581	.132	4.03
1326	335	374	113	366	162.7	.867	.139	4.22
1327	420	391	119	365	161.6	1.13	.138	4.22
1328	443	406	127	367	163.9	1.55	.136	4.24
1329	262	238	98	320	91.1	.384	.346	10.33
1330	310	279	100	320	91.1	1.19	.347	10.59
1331	366	327	102	324	94.5	1.54	.361	10.71
1332	290	266	91	336	112.2	.879	.265	8.07
1333	353	323	103	338	113.4	1.21	.267	8.15
1335	397	369	105	337	114.0	1.54	.266	8.13
1340	271	254	77	323	95.0	.801	.153	[illegible]

TABLE I. - Concluded. SINGLE- AND TWO-PHASE HEAT-TRANSFER DATA AT STATION 11

Run	Temperature: Outside, T_o, °F	Temperature: Inside, T_i, °F	Temperature: Bulk, T_b, °F	Temperature: Saturation, T_s, °F	Pressure, p, lb/sq in. abs	Heat flux, q, Btu/(sec)(sq in.)	Mass-flow rate, $\dot{m}$, lb/sec	Bulk velocity, V_b, ft/sec
1341	347	325	84	324	96.1	0.871	0.191	5.61
1342	376	346	89	325	96.5	1.181	.191	5.61
1343	407	368	93	327	97.3	1.557	.195	5.94
1345	316	300	79	331	104.9	.601	.150	4.56
1346	369	347	92	331	104.9	.868	.141	4.29
1347	398	366	97	333	107.5	1.177	.144	4.39
1348	424	386	104	333	107.7	1.556	.142	4.33
1349	241	224	72	269	40.5	.603	.232	7.04
1350	306	283	81	269	40.4	.875	.228	6.94
1352	331	300	87	269	40.5	1.210	.233	7.09
1353	370	331	89	271	41.6	1.574	.233	7.09
1354	276	260	82	283	51.0	.603	.174	5.29
1355	312	289	81	289	54.2	.873	.183	5.57
1356	348	318	103	272	42.5	1.182	.186	5.68
1358	377	338	109	275	45.0	1.542	.190	5.81
1359	312	296	96	277	45.9	.603	.144	4.39
1360	340	317	104	276	46.3	.873	.144	4.40
1361	364	333	112	275	45.0	1.215	.138	4.22
1362	387	348	120	275	45.1	1.562	.138	4.22
1366	288	265	91	282	49.3	.873	.253	7.71
1367	329	299	98	283	50.4	1.190	.244	7.44
1368	370	331	104	284	52.1	1.570	.256	7.81
1369	326	304	98	298	64.1	.864	.182	5.58
1370	346	315	102	298	64.1	1.180	.186	5.68
1371	386	347	108	298	64.1	1.560	.186	5.68
1372	300	284	94	306	72.2	.596	.144	4.39
1374	335	312	99	303	70.6	.854	.149	4.54
1375	370	341	110	300	68.7	1.190	.144	4.40
1376	399	361	120	304	71.6	1.540	.151	4.63
1377	281	258	107	297	60.7	.854	.281	8.58
1378	345	314	113	297	60.7	1.180	.278	8.50
1379	377	338	118	298	61.3	1.550	.283	8.66
1380	276	260	108	304	71.6	.605	.185	5.65
1381	344	322	113	305	72.7	.862	.185	5.66
1382	364	335	118	305	76.7	1.160	.189	5.79
1383	398	359	123	304	72.2	1.550	.185	5.67
1384	310	295	111	312	81.3	.594	.143	4.37
1385	350	328	112	314	82.5	.855	.151	4.62
1394	396	367	96	342	121.1	1.160	.152	4.63
1395	429	391	109	344	123.6	1.570	.144	4.40
1396	288	256	82	360	152.5	1.200	.351	10.68
1397	338	297	89	353	140.2	1.560	.360	10.95
1398	347	315	88	348	129.2	1.220	.259	7.88
1399	394	355	95	343	122.8	1.570	.256	7.80
1400	340	317	100	368	168.7	.882	.190	5.80
1401	406	379	107	370	172.1	1.180	.194	5.92
1402	437	399	108	371	175.8	1.570	.196	5.99
1404	320	304	98	363	156.8	.602	.129	3.93
1406	394	373	105	358	148.3	.863	.138	4.21
1407	420	390	113	358	148.3	1.210	.144	4.40

Run	Temperature: Outside, T_o, °F	Temperature: Inside, T_i, °F	Temperature: Bulk, T_b, °F	Temperature: Saturation, T_s, °F	Pressure, p, lb/sq in. abs	Heat flux, q, Btu/(sec)(sq in.)	Mass-flow rate, $\dot{m}$, lb/sec	Bulk velocity, V_b, ft/sec
1409	442	404	121	356	148.7	1.560	0.141	4.32
1412	307	284	86	265	37.7	.696	.226	6.94
1413	345	315	90	271	42.4	1.160	.233	7.10
1414	376	336	93	272	43.2	1.550	.235	7.16
1415	343	321	89	275	45.1	.663	.174	5.30
1416	369	337	94	275	45.3	1.250	.174	5.30
1417	387	349	98	276	46.6	1.550	.174	5.31
1418	350	329	93	289	56.4	.615	.127	3.87
1420	374	345	101	288	55.1	1.160	.120	3.66
1421	396	357	112	288	55.1	1.550	.124	3.73
1422	354	315	83	320	89.1	1.550	.363	11.04
1423	399	370	93	333	116.3	1.170	.187	5.70
1424	423	385	98	333	116.3	1.550	.187	5.71
1425	413	384	97	336	123.9	1.180	.160	4.88
1426	437	399	109	339	124.3	1.56	.133	4.06
1427	405	367	87	346	145.2	1.56	.266	8.09
1428	421	383	106	358	147.6	1.16	.186	5.68
1429	392	353	114	357	147.3	1.56	.194	5.94
1430	404	376	115	359	151.5	1.17	.132	4.04
1431	438	400	130	361	154.5	1.57	.132	4.05
1433	338	308	102	277	46.6	1.17	.246	7.51
1435	362	323	108	280	48.7	1.54	.249	7.61
1436	347	316	109	293	59.4	1.17	.175	5.35
1438	379	340	119	290	56.4	1.56	.175	5.36
1439	358	329	115	295	62.8	1.17	.132	4.04
1440	393	363	110	369	172.8	1.20	.186	5.68
1441	430	392	115	369	172.4	1.55	.186	5.69
1442	415	366	114	374	181.7	1.18	.135	4.13
1443	447	409	126	373	180.4	1.55	.132	4.05
1602	170	156	87	316	85.0	.392	.347	10.55
1603	228	208	90	317	86.8	.723	.351	10.68
1604	311	279	95	317	87.6	1.223	.354	10.78
1605	366	327	99	319	88.4	1.567	.355	10.83
1606	195	183	88	336	110.9	.399	.243	7.39
1607	245	228	90	336	110.8	.507	.242	7.37
1608	309	255	95	338	111.5	.910	.246	7.49
1609	377	346	101	337	113.0	1.231	.246	7.51
1610	406	369	106	337	111.9	1.580	.246	7.51
1611	212	201	89	342	121.3	.391	.200	6.08
1612	275	258	94	342	121.3	.619	.200	6.09
1613	350	327	99	343	122.3	.904	.200	6.10
1614	403	373	106	343	122.5	1.229	.200	6.10
1615	428	389	113	344	123.3	1.577	.200	6.09
1616	262	239	105	354	143.3	.881	.356	10.86
1617	315	283	106	354	143.9	1.230	.361	11.01
1618	360	313	109	354	144.5	1.582	.362	11.06
1619	259	242	103	353	142.7	.624	.232	7.09
1620	313	283	106	349	131.9	.909	.243	7.41
1621	374	344	111	349	132.5	1.200	.241	7.36
1622	424	386	117	348	133.1	1.575	.245	7.50

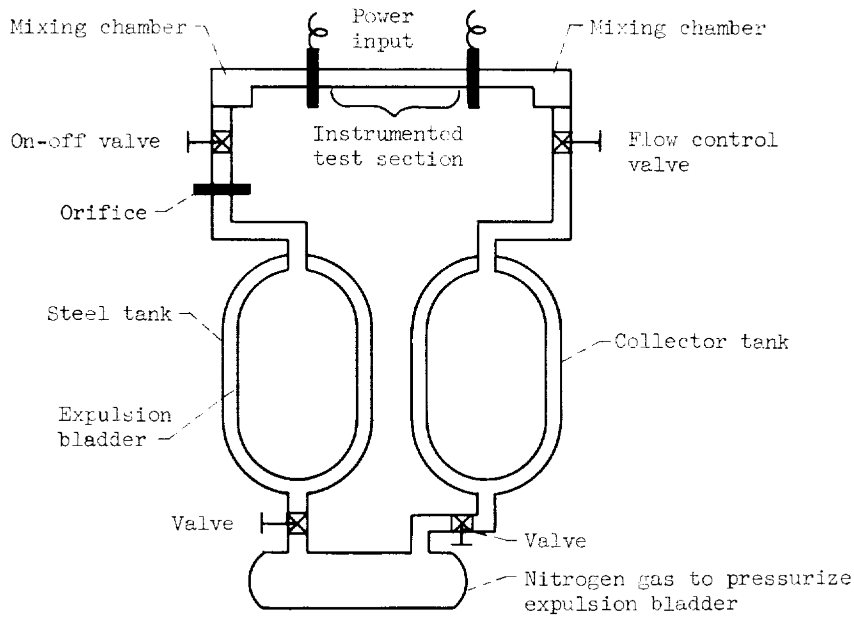
TABLE II. - EXPERIMENTAL DATA FROM VARIOUS SOURCES

Fluid	Pressure, p, lb/sq in. abs	Heat flux, q, Btu/(sq in.)(sec)	Velocity, V, ft/sec	Subcooling, $^{\circ}F$	Number of boiling points	Reference
Water	37 to 179	0.37 to 1.60	3.8 to 12.5	180 to 263	103	Present study
	69 to 307	3.5 to 56.0	73 to 204	197 to 335	18	14
	16 to 202	0.73 to 2.80	6.2 to 12.6	100 to 258	40	11
	2000	0.27 to 1.41	2.4 to 9.5	12 to 148	28	3
	2000	1.9 to 4.9	20.0	116 to 256	16	5
	500	0.026 to 0.85	1.33	6 to 179	30	4
	1500	0.052 to 0.57	1.40	37 to 336	20	4
	2000	0.145 to 0.62	1.40	106 to 282	20	4
Liquid ammonia	170 to 1174	0.38 to 9.00	3.0 to 85.0	37 to 187	31	10

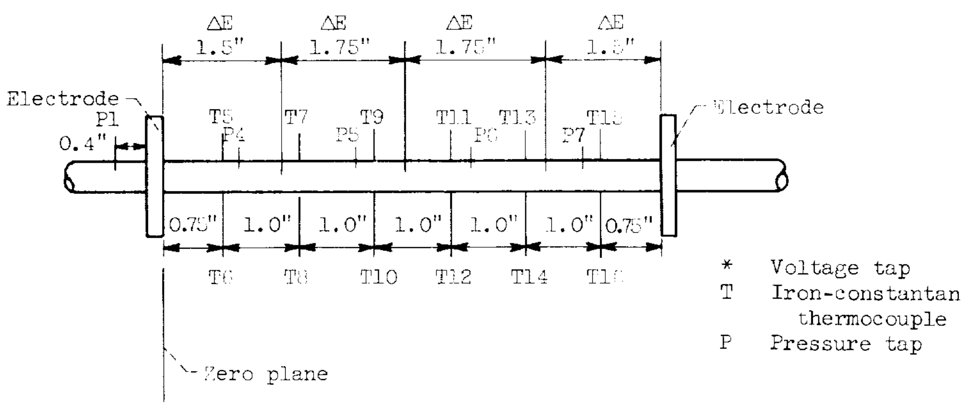
TABLE III. - COMPARISON OF COMPUTED AND EXPERIMENTAL INCIPIENT BOILING POINTS

Run	Pressure, p, lb/sq in. abs	Velocity, V, ft/sec	Experimental wall temperature, $T_{w,exp}$, $^{\circ}F$	Wall temperature calculated by eq. (B1)[a] $T_{w,calc}$, $^{\circ}F$
1359	46	4.39	296	292
1271	100	4.32	349	343
1406	148	4.21	373	371

[a]Laminar thickness ratio δ/C_3 of 1482 μin. at velocity of 4.25 ft/sec (obtained from run 1346).



(a) Flow system.



(b) Instrumentation.

Figure 1. - Schematic drawing of test apparatus.

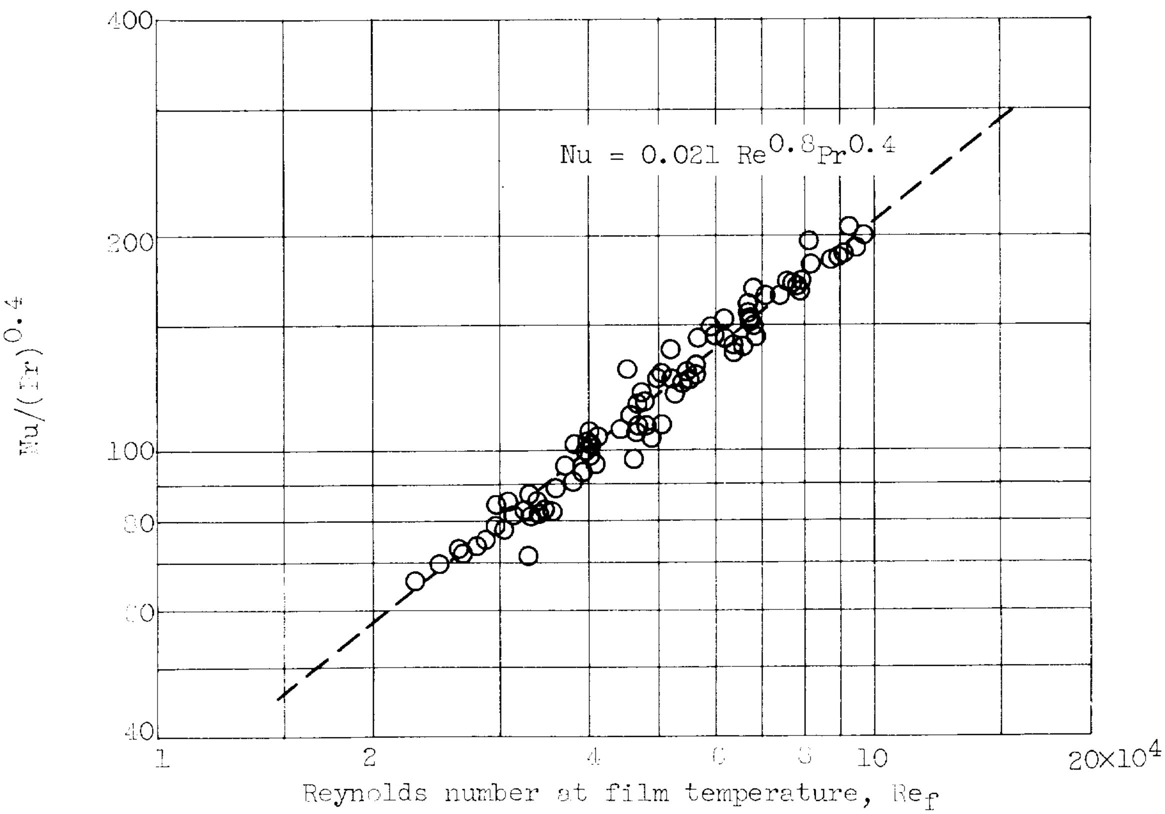


Figure 2. - Correlation of nonboiling heat-transfer data.

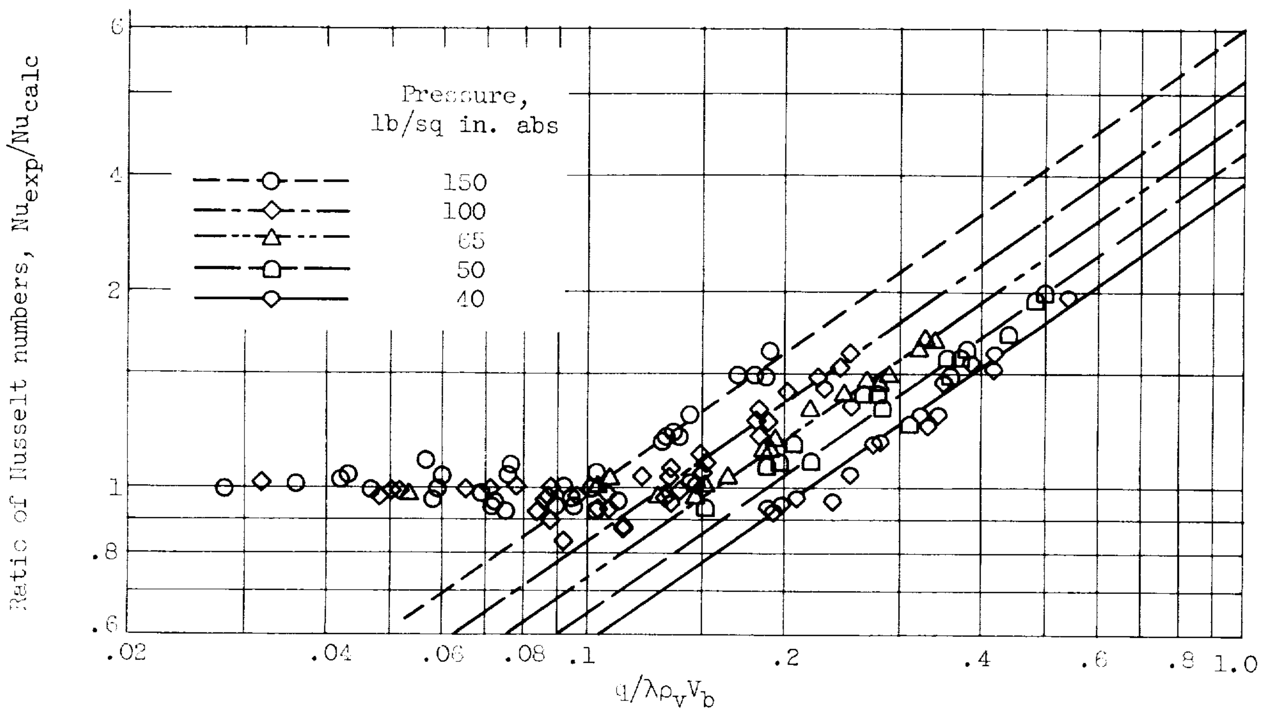


Figure 3. - Partial correlation of boiling heat-transfer data showing pressure effect. Dashed lines drawn at a slope of 0.7.

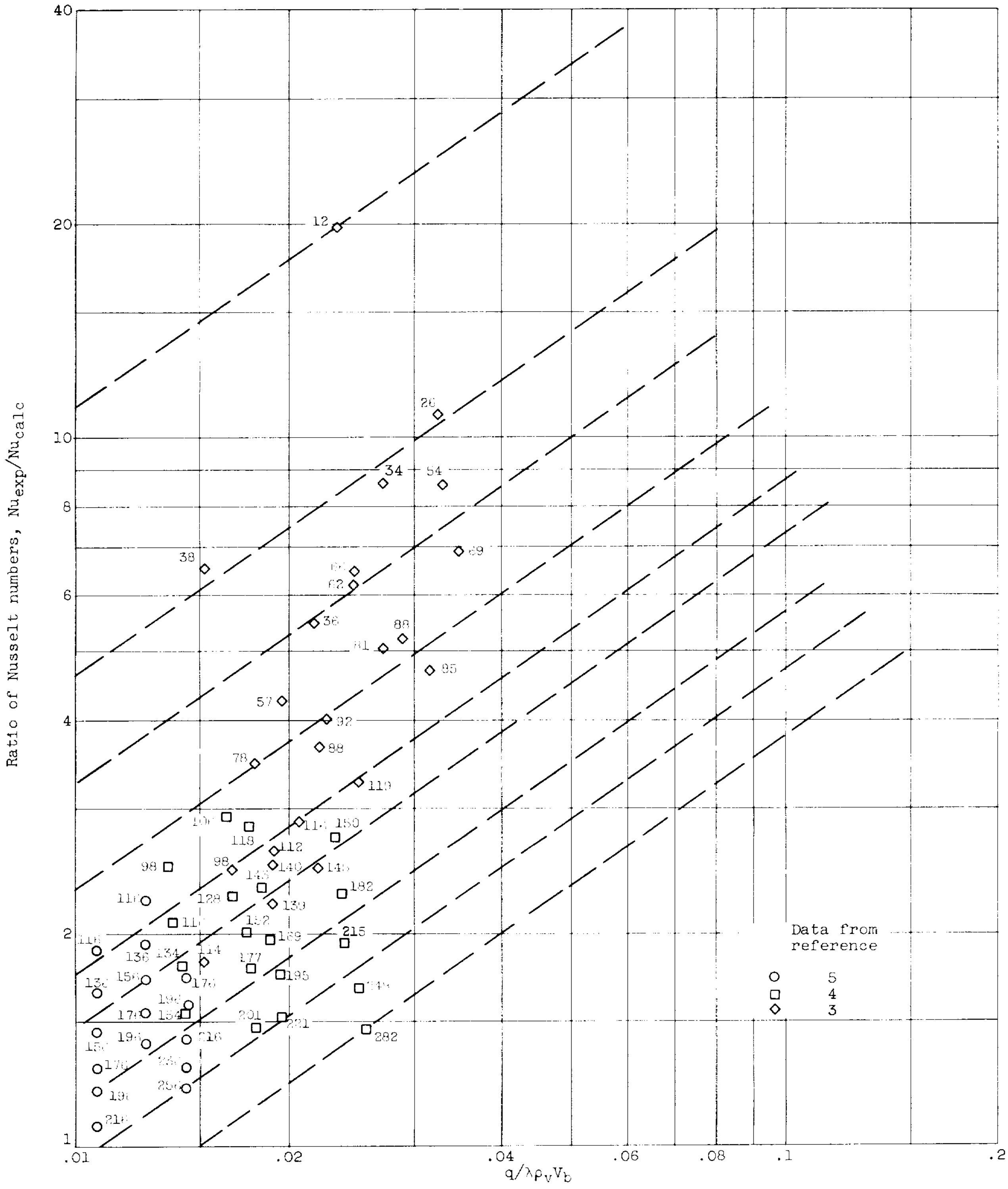


Figure 4. - Effect of subcooling at constant pressure of 2000 pounds per square inch absolute. Degrees of subcooling indicated next to data points. Dashed lines drawn at a slope of 0.7.

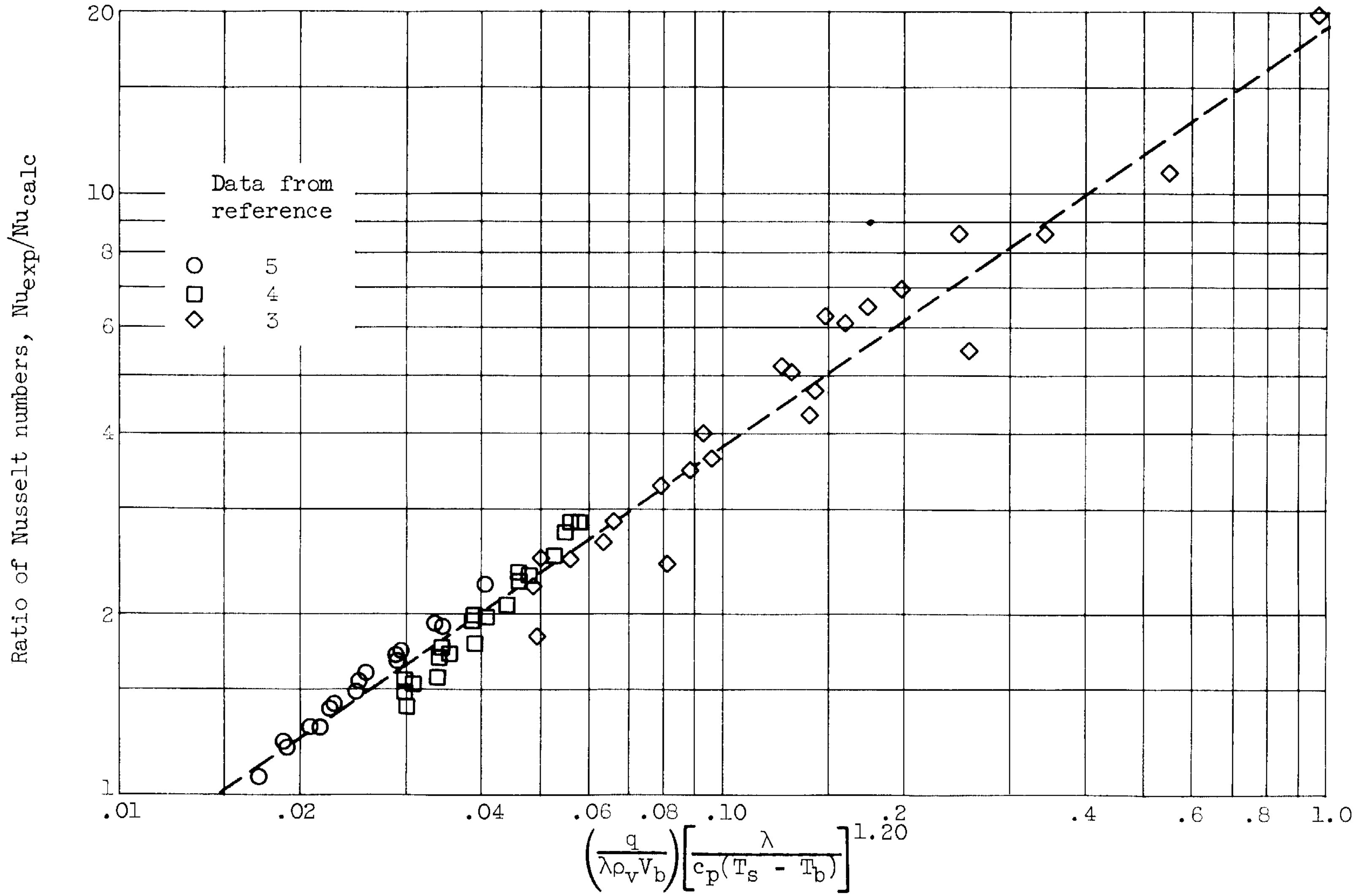


Figure 5. - Effect of subcooling compensated by a correcting parameter. (Replot of data presented in fig. 4.) Constant pressure, 2000 pounds per square inch. Dashed line drawn at a slope of 0.7.

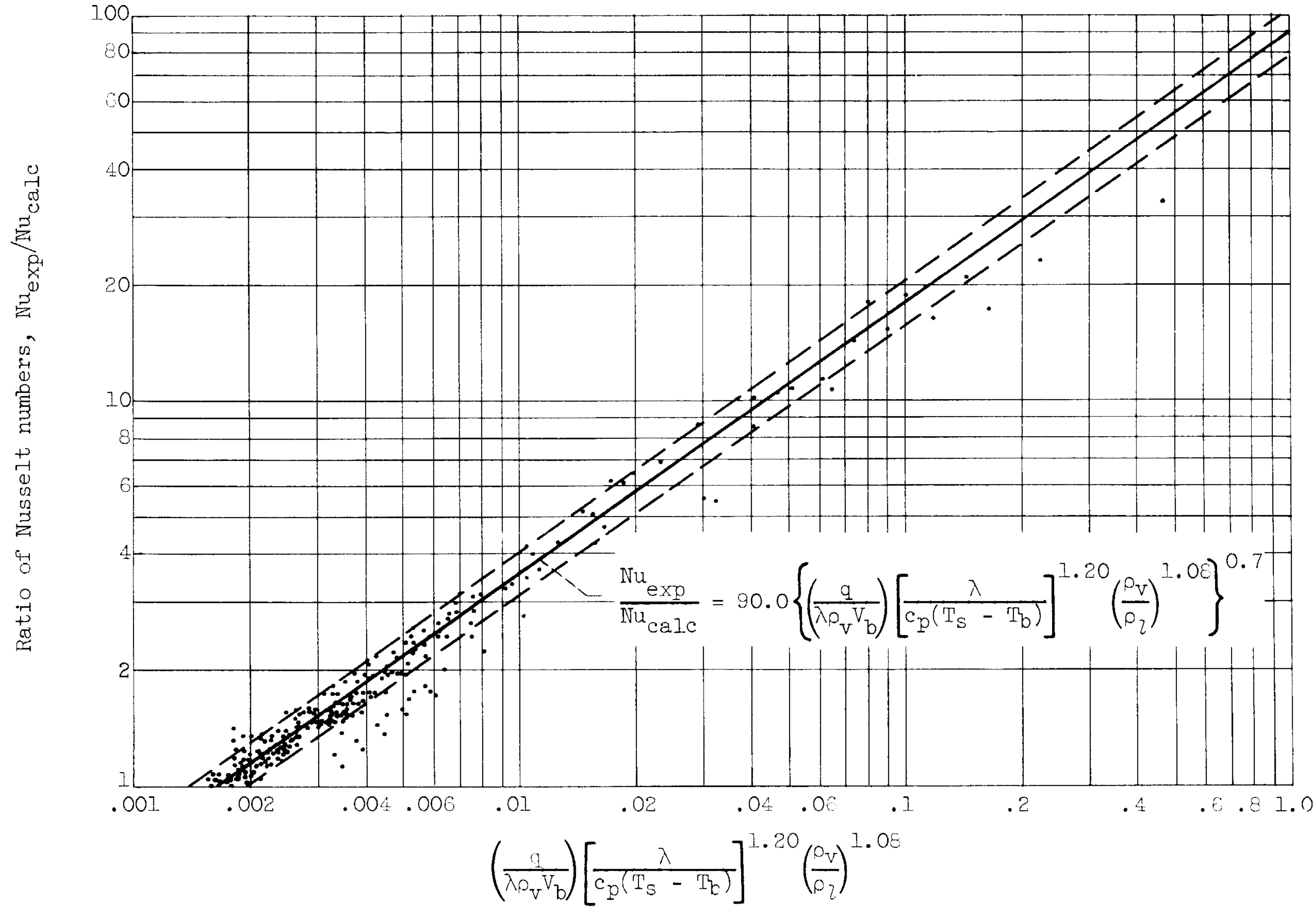


Figure 6. - Completed correlation using density-ratio parameter to compensate for remaining pressure effect. Includes all boiling-water data from present investigation and references 3, 4, 5, 11, and 14.

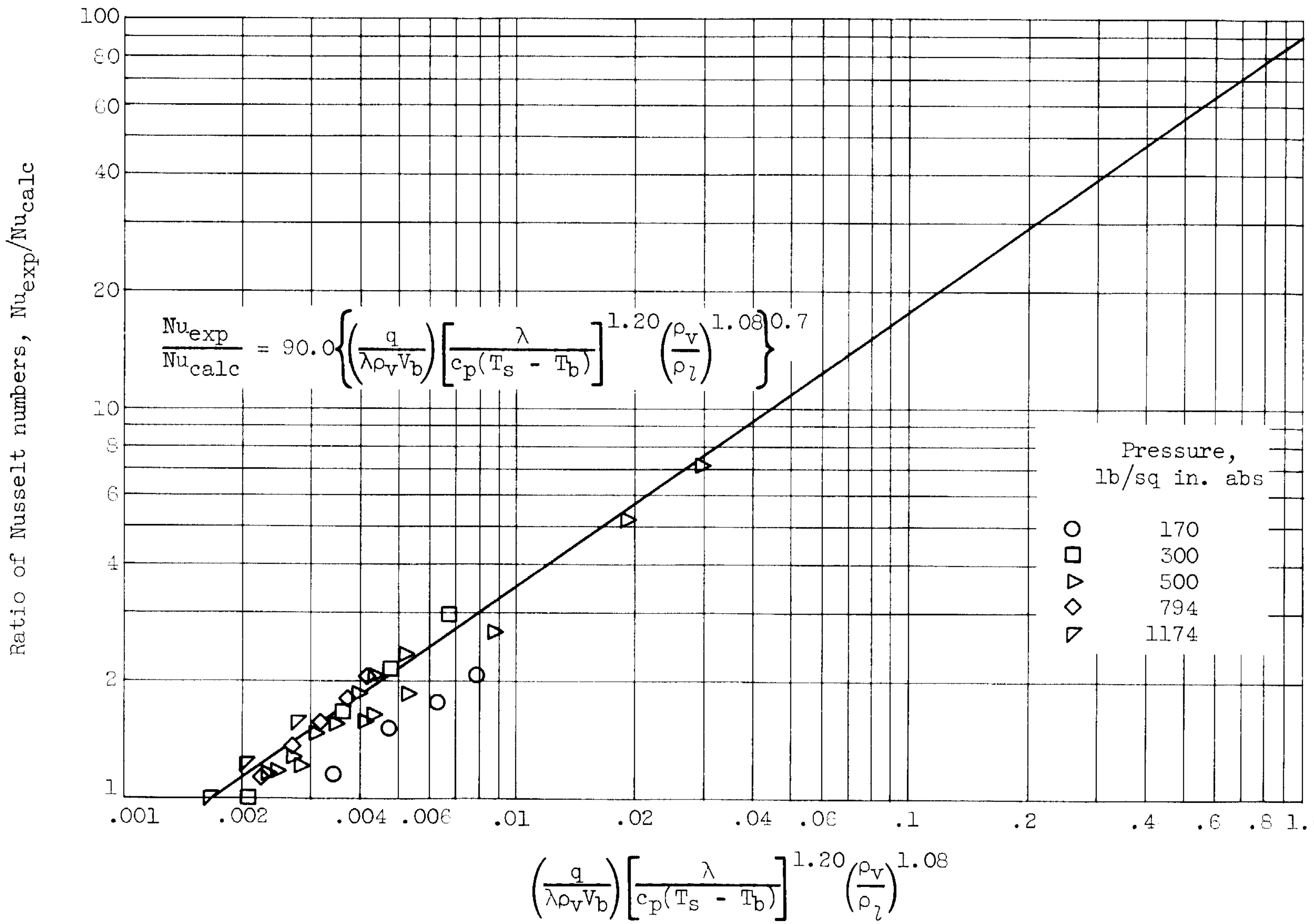


Figure 7. - Correlation of liquid-ammonia data from reference 10. Range of pressure, 170 to 1174 pounds per square inch absolute.

NASA-Langley, 1963 E-1567